

Energetic optimization of the performances of a hot air engine for micro-CHP systems working with a Joule or an Ericsson cycle

M. Creyx^{a,b,c,*}, E. Delacourt^{a,b}, C. Morin^{a,b}, B. Desmet^{a,b}, P. Peultier^c

^a Université Lille Nord de France, F-59000 Lille, France

^b UVHC, TEMPO, Le Mont Houy, F-59313 Valenciennes Cedex 9, France

^c Enerbiom, Ruche du Hainaut, 350 rue Arthur Brunet, F-59721 Denain, France

ARTICLE INFO

Article history:

Received 28 March 2012

Received in revised form

11 September 2012

Accepted 28 October 2012

Available online 14 December 2012

Keywords:

Ericsson engine

Joule and Ericsson cycles

Thermodynamic optimization

Engine performances

Cogeneration

Hot air engines

ABSTRACT

The micro combined heat and electrical power systems (micro-CHP) with hot air engines are well adapted for solid biomass upgrading, in particular, the Ericsson engines working with an open cycle and an external combustion. This paper presents a model of an Ericsson engine with a compression and an expansion cylinder which allows a thermodynamic optimization of the engine performances in a global approach. A sensitive analysis on the influent parameters is carried out in order to determine the optimal working conditions of the engine: temperature and pressure range, expansion cycle shape with a late intake valve closing or an early exhaust valve closing, heat transfers through the wall of the cylinders. This study, focused on thermodynamic aspects, is a first step in the design of an Ericsson engine.

© 2012 Elsevier Ltd. All rights reserved.

1. Introduction

Due to the increasing costs of fossil fuels, the interest in small-scale combined heat and electrical power systems (CHP) working with renewable energies is rising. In particular, biomass micro-CHP systems are emerging in a range of electrical power from 1 to 50 kW [1,2]. Our work is focused on solid biomass upgrading through micro-cogeneration systems with external combustion.

Several cogeneration technologies can be used in biomass micro-CHP plants: the micro gas turbines, the internal combustion engines, the vapour turbines using an Organic Rankine Cycle (ORC) and the hot air engines. The micro gas turbines, working with a direct combustion or a gasification of biomass and with an internal or external combustion, present a low efficiency for the range of power considered, with fouling problems [3]. The internal combustion engines can be used only with liquid or gaseous

biomass, the efficiency depending strongly on the combustion stability. The ORC systems, which are external combustion engines, require a stable heat source [4]. The hot air engines are currently used in biomass cogeneration [1]: they can work with gasification or direct combustion of solid biomass and the combustion can be internal or external. In the case of an external combustion, the thermodynamic cycle of the fluid can be open or closed. When an open cycle is used, the heat evacuated with the exhaust gas can be reused in other parts of the cogeneration (for example in a pre-heater or in a regenerator), which enhances the global performances. The closed cycles face strong technological stress to ensure tightness and require the presence of a cold sink. Different hot air engine technologies can be used: the Stirling engine which works with a closed Stirling cycle, and the Ericsson engine which can work with an open or closed Joule or Ericsson cycle (respectively cycle with isentropic – adiabatic reversible – transformations and cycle with isothermal transformations).

Several Stirling cycle engines have been developed for biomass upgrading: Biedermann et al. [5] tested a β -type Stirling engine, while Thiers et al. [6] and Podesser [7] described α -type Stirling engines designed for biomass micro-CHP. The Ericsson engines were never used for biomass cogeneration, despite their advantages. They can be used with open cycles and according to Bonnet et al. investigations [8], the heat exchanger design does not require

* Corresponding author. Laboratoire TEMPO, Université de Valenciennes et du Hainaut-Cambrésis, Le Mont Houy, 59313 Valenciennes Cedex 09, France. Tel.: +33 3 27 51 19 83; fax: +33 3 27 51 19 61.

E-mail addresses: creyx@enerbiom.com (M. Creyx), eric.delacourt@univ-valenciennes.fr (E. Delacourt), celine.morin@univ-valenciennes.fr (C. Morin), bernard.desmet@univ-valenciennes.fr (B. Desmet), peultier@enerbiom.com (P. Peultier).

Nomenclature		Greeks symbols	
c_p	air specific heat capacity at constant pressure, J/kg.K	α_{in}	air intake setting coefficient
c_v	air specific heat capacity at constant volume, J/kg.K	α_{EEVC}	setting coefficient of residual mass compression
$\dot{E}_x$	exergy flow rate, W	γ	isentropic coefficient
h	specific enthalpy of air, J/kg	η_{th}	thermodynamic efficiency, %
IMP	indicated mean pressure, Pa	η_{ex}	exergetic efficiency, %
k	polytropic coefficient	T	torque, Nm
m	mass, kg	τ	volume ratio $(V_M - V_m)/V_m$
m_{cycle}	intake mass per cycle, kg/cycle	ω	rotational speed, rad/s
N	rotational speed, rpm	Subscripts	
p	pressure, Pa	atm	atmospheric
P_i	theoretical indicated power, W	C	compression cylinder
P_{th}	thermal power, W	c	compression cycle
Q_{eff}	effective heat transferred, J	E	expansion cylinder
q_m	air mass flow rate, kg/s	E	Ericsson cycle
q_m^*	adimensional air mass flow rate	e	expansion cycle
r	specific gas constant, J/kg K	EEVC	early exhaust valve closing
s	specific entropy of air, J/kg K	H	heat exchanger
S	entropy, J/K	h	heat exchanger outlet
T	temperature, K	in	intake
V	volume, m ³	J	Joule cycle
V_{swept}	cylinder swept volume, m ³	$M, \max$	maximum
W_i	theoretical indicated work, J	$m, \min$	minimum
w_i	theoretical specific indicated work, J/kg	out	engine outlet
W_{ij}	work received by the piston from step i to step j , J	r	residual
		ref	reference

such a restricting compromise between a large area and a small volume as the Stirling engines [9] and the investment is potentially cheaper than for a Stirling engine (lower working pressures). Kaushik et al. [10] showed that the Ericsson engine performances are comparable to the Stirling engine performances and Wojewoda et al. [11] demonstrated that the Ericsson engine performances can be better in some cases of closed cycles presented in their works.

The Ericsson engines can work under two cycle types: isentropic compressions and expansions matching a Joule cycle (Ericsson engine working with a Joule cycle), or isothermal compressions and expansions matching an Ericsson cycle (Ericsson engine working with an Ericsson cycle). The working conditions of the compression and expansion cylinders can be different: an Ericsson engine can have a compression cylinder working with a Joule cycle and an expansion cylinder working with an Ericsson cycle. The inverse configuration of the working conditions (Joule expansion cycle and Ericsson compression cycle) can also exist.

Several numerical studies [11–15], detailed in Table 1 with their engine configurations and their working conditions, have

investigated the performances of an Ericsson engine composed of two cylinders for the compression and expansion cycles. The closed Ericsson cycle engine with regenerator has been described by Kaushik et al. [10] and Chen et al. [16] in finite-time thermodynamic models including the heat and regenerative losses and the cycle irreversibilities. These studies have led to an optimized design of the engine power output and efficiency. Dynamic models of Ericsson engines working with a Joule cycle have been set up by Wojewoda et al. [11], who have worked on a closed cycle, and by Lontsi et al. [14], who have modelled an open cycle. Wojewoda et al. have shown the influence of the engine configuration on the performances. Lontsi et al. have demonstrated the engine working stability. Moss et al. [12] have proposed a design of an Ericsson internal combustion engine working with an open Joule cycle from a steady state thermodynamic model including the effects of frictional losses. Bell et al. [13] have designed a similar engine, giving the optimal working conditions. A thermodynamic model developed by Touré [15] describes an Ericsson external combustion engine working with an open Joule cycle in a steady state, including

Table 1
Numerical studies from the literature: Ericsson engine configuration and working conditions.

Investigator(s)	Ericsson engine configuration	Working conditions			
		p_h/p_{min}	T_{max}	N	V
Wojewoda et al. [11]	Pre-heater, heater and cooler External combustion – Closed Joule cycle	4.75	873–1273 K	500–3000 rmp	
Moss et al. [12]	Recuperator Internal combustion – Open Joule cycle	1–16	1000–1200 K	300–2000 rpm	$V_{swept,e} = 4650 \text{ cm}^3$
Bell et al. [13]	Pre-heater and combustion chamber Internal combustion – Open Joule cycle	1–20	1300 K		$V_{swept,e}/V_{swept,c} = 1.4–4.6$
Lontsi et al. [14]	Heat exchanger External combustion – Open Joule cycle	3.6	873 K	480 rpm	$V_{Me} = 4885 \text{ cm}^3$ $V_{me} = 362 \text{ cm}^3$
Touré [15]	Heat exchanger External combustion – Open Joule cycle	2.5	873 K	950 rpm	$V_{Me} = 650 \text{ cm}^3$ $V_{me} = 10 \text{ cm}^3$

a sensitive analysis of the working parameters (maximal pressure, temperature, volume ratio and swept volume ratio of the compression and expansion cylinders, air mass flow rate).

This paper presents a theoretical thermodynamic optimization of an Ericsson engine working with an open Joule or Ericsson cycle, which is a micro-CHP system well adapted to the solid biomass upgrading. The engine is composed of a compression cylinder (C), an expansion cylinder (E) and a heat exchanger (H) as shown in Fig. 1. This study is necessary to design an Ericsson engine with optimized working conditions, in a global approach. The compression and expansion cylinders are coupled to show the influence of working conditions on the global engine performances. Indeed, the separated optimization of the compression and expansion cylinders is not always linked with the optimal working conditions of the coupled system. The engine configuration is first described, followed by the theoretical model. Then, the results of the computer simulation in a steady state are presented, including a comparison with available literature data in a limited range and a sensitive analysis of the thermodynamic parameters. Finally, the optimal engine configuration is discussed.

2. Description of the Ericsson engine configuration investigated

The Ericsson engine studied, working with an open cycle, is composed of two cylinders: a compression (C) and an expansion (E) cylinder, and a heat exchanger (H) (cf. Fig. 1). The working fluid is air which follows two thermodynamic cycles: the first in the compression cylinder and the second in the expansion cylinder. The air enters into the compression cylinder at atmospheric pressure and ambient temperature, is then compressed, then sent into the heat exchanger where it is heated to reach a fixed temperature (defined as a parameter), enters then into the expansion cylinder where it expands and is finally rejected in the atmosphere. The real pressure reached when the engine functioning becomes stable (pressure between the compression cylinder outlet and the expansion cylinder inlet) depends on the volumes of the compression and expansion cylinders, on the heat exchanger outlet temperature and on the timing of the valves closing. The heat transferred to the working air in the heat exchanger is adjusted in the model to obtain a determined temperature at the heat exchanger outlet.

No regenerator is taken into account in this engine, even if it could enhance the performances of the system. Indeed, the

regenerator improves the cogeneration efficiency but does not change the heat quantity required by the engine. The optimal efficiencies obtained with our model are minimum values which can be improved by adding modifications to the global cogeneration configuration, for example a regenerator or a pre-heater.

The global engine optimization, taking into account the coupling of the thermodynamic transformations of the working air in the compression and expansion cylinders, is essential to highlight a realistic range of optimal working conditions, that is investigated here.

The shapes of the compression and expansion cycles can be modified to enhance the engine performances, as shown in the P–V diagrams in Figs 2 and 3. Two cycle types can be chosen: a Joule or an Ericsson cycle. The first cycle consists of isentropic compressions and expansions in both cylinders, i.e. perfectly insulated cylinder walls. The second one includes isothermal compressions and expansions in both cylinders, i.e. cylinder walls with infinite heat conductivity. The real compression and expansion cycles are polytropic, with a polytropic coefficient close to the isentropic coefficient γ , so that the real performances are situated between the Ericsson and Joule cycle performances, with values closer to the Joule cycle performances. The comparison between Ericsson engines working with Joule and Ericsson cycles shows the effect of heat transfers in the cylinders walls on the engine performances.

Adjustments of Ericsson and Joule cycle shapes in the expansion cylinder can be added. The intake phase can be lengthened, so that the expansion is incomplete and a pressure balance occurs when the exhaust valve is opened, as represented in Fig. 3 with the cycle shape $0_e-1_e-2'_e-3'_e-3''_e$. The exhaust phase can be shortened, with an early exhaust valve closing and a compression of the residual mass present in the expansion cylinder (cf. cycle shape $0'_e-0''_e-1_e-2_e-3_e$ in Fig. 3). These both adjustments can be coupled, with a cycle shape $0'_e-0''_e-1_e-2'_e-3'_e-3''_e$ (cf. Fig. 3). The Ericsson and Joule cycle modifications, not studied in the literature, are modelled in this paper.

3. Theoretical model

3.1. Description

The open cycle Ericsson engine model is based on the thermodynamic transformations of the working air mass in a steady state. The air is considered as an ideal gas. The dynamic effects in transient phases, the mechanical frictions, and the pressures drops in the distribution system and in the heat exchanger are not

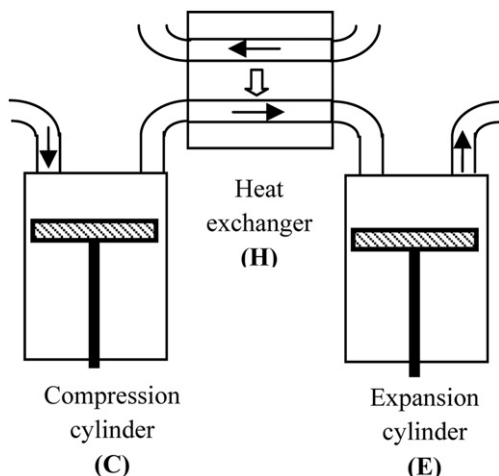


Fig. 1. Engine configuration.

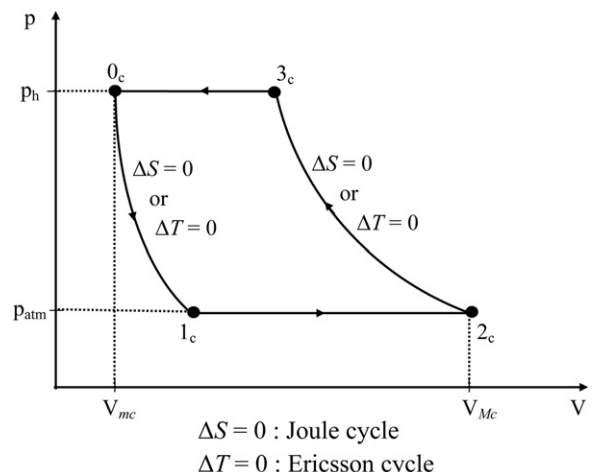


Fig. 2. P–V diagram of compression cycle.

An energy balance on intake air mass m_{in} and mass m_1 in the cylinder considered (here, $m_1 = m_{1c}$ for the compression cylinder and $m_1 = m_{1e}$ for the expansion cylinder), under the hypothesis of an adiabatic mass transfer valid for both Joule and Ericsson cycles, gives the temperatures T_{2c} and T_{2e} after air intake:

$$T_j = (m_1 T_i + m_{in} T_k) / (m_1 + m_{in}) \quad (6)$$

The work transferred during the air intake is:

$$W_{ij} = m_1 r T_i - (m_{in} + m_1) r T_j \quad (7)$$

3.2.3. Expansion cylinder air intake: pressure balance

The pressure balance for the expansion cylinder air intake, transformation ($i-j$), matches the transformations (0_e-1_e) or ($0''_e-1_e$) according to Fig. 3. The air mass transfer is supposed adiabatic for both Joule and Ericsson cycles. The energy balance on air intake mass m_{ij} and residual mass m_{re} gives the air intake mass m_{ij} entering the expansion cylinder during the pressure balance and the mass in the expansion cylinder after the pressure balance m_{1e} :

$$m_{ij} = (p_h - p_i) V_{me} / (\gamma r T_h) \quad (8)$$

$$m_{1e} = m_{re} + m_{ij} \quad (9)$$

The expansion cycle shape with air intake pressure balance leads to the following equation of the mass per cycle entering each cylinder:

$$m_{cycle} = m_{in} + m_{ij} \quad (10)$$

This last equation results from the two intake phases of the expansion cycle: pressure balance (0_e-1_e) or ($0''_e-1_e$) in Fig. 3 and air displacement (1_e-2_e) or ($1_e-2'_e$) in Fig. 3. The mass of air entering the expansion cylinder during the pressure balance depends on the time when the exhaust valve is closed at point $0'_e$ (early exhaust valve closing adjustment). Indeed, an early exhaust valve closing induces a compression of the air residual mass in the cylinder at point $0''_e$, which sets the pressure level in the cylinder at the beginning of the pressure balance. The mass entering the expansion cylinder during the phase of air displacement is linked to the duration of this phase which depends on the time when the intake valve is closed at point $2'_e$ in Fig. 3 (late intake valve closing adjustment).

3.2.4. Exhaust of the compression and expansion cylinders

The air exhaust, process ($i-j$) matching the transformations (3_c-0_c) in the compression cylinder (cf. Fig. 2), or (3_e-0_e), ($3''_e-0_e$), ($3_e-0'_e$) or ($3''_e-0'_e$) in the expansion cylinder (cf. Fig. 3), depending on the presence of complete or incomplete expansion, or the presence of an early exhaust valve closing, is considered isobaric and isothermal for both Joule and Ericsson cycles. The residual mass is evaluated with the ideal gas law. The transferred work is:

$$W_{ij} = (m_i - m_j) r T_i \quad (11)$$

3.2.5. Expansion cylinder exhaust: pressure balance

This process matches the transformation ($3'_e-3''_e$) of the expansion cycle P–V diagram (cf. Fig. 3). The air mass initially present in the cylinder at intermediate pressure expands in the exhaust pipe with a globally adiabatic process for both Joule and Ericsson cycles, reaching the atmospheric pressure. The air remaining in the cylinder at point $3''_e$ is supposed to expand with an isentropic transformation. Eq. (2) with $i = 3'_e$, $j = 3''_e$ and $k = \gamma$ links the temperature at point $3'_e$ to the pressure ratio occurring during the expansion.

When a pressure balance occurs at the exhaust step of the expansion cylinder, the temperature of the exhaust pipe T_{out} is different of the temperature in the cylinder after exhaust step T_i , with $i = 0_e$ or $i = 0'_e$ (cf. Fig. 3). An energy balance at the point $3'_e$ on the working air mass which expands in the cylinder and in the exhaust pipe gives the exhaust pipe temperature:

$$T_{out} = \frac{p_{atm}(V_{Me} - V_i) + m_{3'_e} c_v T_{3'_e} - m_{re} c_v T_i}{(m_{3'_e} - m_i) c_p} \quad (12)$$

3.3. Equations of global performance parameters

The reference conditions chosen are the atmospheric conditions, in particular concerning the exergy calculations [17].

3.3.1. Effective heat transferred to the working air

The effective heat transferred to the working air for an Ericsson engine including Ericsson compression and expansion cycles is:

$$Q_{eff} = \underbrace{m_{cycle} c_p (T_h - T_{0_e})}_{Q_{eff,H}} + \underbrace{(m_{cycle} + m_{re}) r T_i \ln \left(\frac{p_i}{p_j} \right)}_{Q_{eff,E}} + \underbrace{m_{re} r T_{0_c} \ln \left(\frac{p_{0_c}}{p_{1_c}} \right)}_{Q_{eff,C}} \quad (13)$$

The transformation ($i-j$) refers to the transformations (2_e-3_e) or ($2'_e-3'_e$) of the expansion cycle (cf. Fig. 3). The term $Q_{eff,H}$ refers to the heat transferred to the working air in the heat exchanger. The term $Q_{eff,E}$ corresponds to the heat transferred through the expansion cylinder wall during the expansion transformation of the Ericsson expansion cycle (cf. steps (2_e-3_e) and ($2'_e-3'_e$) in Fig. 3). The term $Q_{eff,C}$ matches the heat transferred during the expansion transformation of the Ericsson compression cycle (cf. step (0_c-1_c) in Fig. 2) through the compression cylinder wall. In the case of an Ericsson engine working with a Joule compression cycle, $Q_{eff,C}$ is equal to 0. In the same way, with a Joule expansion cycle, $Q_{eff,E}$ is equal to 0.

3.3.2. Theoretical indicated work and theoretical specific indicated work

The theoretical indicated work corresponds to the indicated work of the thermodynamic cycle of the working air recovered by the driving shaft. It is evaluated with the following equation:

$$W_i = |W_{ie}| - W_{ic} \quad (14)$$

with W_{ie} : indicated work of the expansion cycle (cf. Fig. 3) and W_{ic} : indicated work of the compression cycle (cf. Fig. 2). The theoretical specific indicated work is defined as the theoretical indicated work divided by the air mass per thermodynamic cycle m_{cycle} entering the engine:

$$w_i = W_i / m_{cycle} \quad (15)$$

3.3.3. Indicated mean pressure

The indicated mean pressure is:

$$IMP = W_i / (V_{Me} - V_{me}) \quad (16)$$

3.3.4. Torque

As the compression and expansion cylinders of the engine are decoupled, one engine thermodynamic cycle per motor's shaft turn is realized. So, the torque equation is:

$$T = W_i/2\pi \quad (17)$$

3.3.5. Air mass flow rate in the engine

The air mass flow rate is only linked to the rotational speed and the intake air mass per cycle in the engine:

$$q_m = m_{\text{cycle}} \cdot \omega / 2\pi \quad (18)$$

This equation is valid for the condition of one cycle per crankshaft revolution.

3.3.6. Thermal power and theoretical indicated power

The thermal power is defined as:

$$P_{\text{th}} = Q_{\text{eff}} \cdot \omega / 2\pi \quad (19)$$

and the theoretical indicated power is defined as:

$$P_i = T \cdot \omega \quad (20)$$

3.3.7. Theoretical thermodynamic efficiency

The thermodynamic efficiency is calculated from the ratio of theoretical indicated work and transferred heat:

$$\eta_{\text{th}} = W_i/Q_{\text{eff}} \quad (21)$$

3.3.8. Exergy flow and efficiency

The exergy flow linked to the heat flux transferred by the heat exchanger is expressed as:

$$\dot{E}x_H = q_m [(h(T_h, p_h) - T_{\text{atm}} s(T_h, p_h)) - (h(T_{0c}, p_{0c}) - T_{\text{atm}} s(T_{0c}, p_{0c}))] \quad (22)$$

The exergy flow at the engine outlet is:

$$\dot{E}x_{\text{out}} = q_m [(h(T_{\text{out}}, p_{\text{atm}}) - T_{\text{atm}} s(T_{\text{out}}, p_{\text{atm}})) - (h(T_{\text{atm}}, p_{\text{atm}}) - T_{\text{atm}} s(T_{\text{atm}}, p_{\text{atm}}))] \quad (23)$$

This exergy flow allows the analysis of the engine performances concerning the exergy recovery and conversion.

The exergetic efficiency is evaluated from the ratio of theoretical indicated power and exergy flow coming from the heat exchanger:

$$\eta_{\text{ex}} = P_i/\dot{E}x_H \quad (24)$$

4. Results

First, the results have been compared with literature references for specific and restricted conditions. Then, several factors influencing the engine performances have been investigated: the heat exchanges in the compression cylinder, the expansion cycle shape, the pressure p_h and the temperature T_h of the heat exchanger.

To simplify the results analysis, the study of the heat exchanges in the compression cylinder has been performed with an arbitrarily

fixed expansion cycle shape: a Joule cycle with complete expansion and without residual mass compression (cf. cycle $(0_e-1_e-2_e-3_e)$ in Fig. 3), which ensures a fixed air mass flow rate in the compression cylinder. In the same way, the influence of the expansion cycle shape, the pressure p_h and the temperature T_h on the engine performances have been studied with a Joule compression cycle (isentropic transformations shown in Fig. 2), as well as for the Ericsson expansion cycles. Indeed, the Joule compression cycle is better adapted for the global engine performances enhancement than the Ericsson compression cycle as developed in the following section.

The isentropic coefficient γ is considered as a constant in the calculations, with a value between 1.35 and 1.4 according to the range of temperatures considered (20–650 °C). This leads to a relative error of less than 3.4% for the thermodynamic efficiency and of less than 7.9% for the indicated mean pressure.

4.1. Comparison of the model with results available in the literature

In Table 2, a comparison between the performances of several Ericsson engines with two cylinders described in the literature (configurations detailed in Table 1) and the performances calculated with our model for the same working conditions is shown for particular cases of engine speed N , temperature T_h and pressure p_h at the heat exchanger outlet.

Our results are similar to the data available in the literature in regard to the efficiency η_{th} , the theoretical indicated power P_i of the global engine, the air mass flow rate q_m , the theoretical specific indicated work w_i and the theoretical indicated power generated only by the expansion cylinder $P_{i,e}$. Some studies [12,13] take into account the frictional effects in the engine. When compared to previous studies [11–15], our model presents satisfactory results. However, the thermal power of the engine described by Moss et al. [12] is lower than the calculated value from our model, which can be explained by the absence of a regenerator in our model. An other difference is the thermal efficiency given by Bell et al. [13], higher than the efficiency obtained with the simulation because of the engine configuration which includes a regenerator and a combustion chamber (internal combustion). When the calculation conditions are close to our model [14,15], the results are very similar.

Our simulation gives the performances of the basic configuration of the Ericsson engine (two cylinders, a heat exchanger) with a focus on the effect of the coupling of the compression and expansion cylinders and on the effect of the modifications of the thermodynamic compression and expansion cycles: isothermal and isentropic transformations, late intake valve closing (incomplete expansion) and early exhaust valve closing (residual mass compression) in the expansion cylinder.

Table 2

Comparison between our model results and literature references.

Investigators	Working condition	Reference results	Our model results
Wojewoda et al. [11]	$p_h/p_{\text{min}} = 95/20$ $T_h = 1273 \text{ K}$ $N = 3000 \text{ rpm}$	$q_m = 0.2 \text{ kg/s}$ $P_i = 28 \text{ kW}$ $\eta_{\text{th}} = 32\%$	$q_m = 0.2 \text{ kg/s}$ $P_i = 28.4 \text{ kW}$ $\eta_{\text{th}} = 33.5\%$
Moss et al. [12]	$p_h/p_{\text{atm}} = 7.5$ $N = 1000 \text{ rpm}$	$P_i = 5 \text{ kW}$ $P_{\text{th}} = 8.12 \text{ kW}$ $\eta_{\text{th}} > 40\%$	$P_i = 5.0 \text{ kW}$ $P_{\text{th}} = 12.1 \text{ kW}$ $\eta_{\text{th}} = 41.5\%$
Bell et al. [13]	$p_h/p_{\text{atm}} = 7$ $T_h = 1300 \text{ K}$	$w_i = 250\text{--}300 \text{ kJ/kg}$ $\eta_{\text{th}} = 40\text{--}50\%$	$w_i = 287 \text{ kJ/kg}$ $\eta_{\text{th}} = 36.59\%$
Lontsi et al. [14]	$p_h/p_{\text{atm}} = 3.6$ $T_h = 873 \text{ K}$ $N = 480 \text{ rpm}$	$\eta_{\text{th}} = 27\%$ $q_m = 0.0165 \text{ kg/s}$	$\eta_{\text{th}} = 28\%$ $q_m = 0.02 \text{ kg/s}$
Touré [15]	$p_h/p_{\text{atm}} = 2.5$ $T_h = 873 \text{ K}$ $N = 950 \text{ rpm}$	$q_m = 0.005 \text{ kg/s}$ $P_{i,e} = 1.1 \text{ kW}$	$q_m = 0.005 \text{ kg/s}$ $P_{i,e} = 1.1 \text{ kW}$

4.2. Influence of heat exchanges in the compression cylinder wall

In order to show the influence of heat exchanges in the compression cylinder wall on the global engine performances, two simulations have been performed with a Joule and an Ericsson compression cycle. The Joule cycle models isentropic transformations in an insulated cylinder wall whereas the Ericsson cycle models isothermal transformations with infinite heat exchanges in the cylinder wall. Several parameters are fixed in the simulations: the pressure ratio, the pressure and temperature of intake air, and the compression cylinder unswept volume. A fixed expansion cycle is defined to ensure an identical intake air mass in the compression cylinder for both cases. The difference between the two simulations is the value of the polytropic coefficient k for compression and expansion in the compression cycle: 1 for Ericsson cycle and γ for Joule cycle (cf. Eqs. (1) and (2)). The results are presented in Table 3.

The compression ratio of the compression cylinder, noted τ_c , is higher for the Ericsson compression cycle because in the P–V diagram, the isothermal residual mass expansion (step (0_c–1_c) in Fig. 2) requires a larger volume difference ($V_{1c}-V_{0c}$) than the isentropic expansion to reach the atmospheric pressure p_{atm} , and the intake phase (step (1_c–2_c) in Fig. 2) requires the same volume difference for both Joule and Ericsson compression cycles. This leads to a maximum compression cylinder volume more important in the case of an Ericsson cycle.

The theoretical specific indicated work of the engine w_i (Eq. (15)) obtained with an isothermal Ericsson compression cycle is higher than the one of the Joule cycle. Indeed, the theoretical indicated work of the compression cycle W_{ic} , which is subtracted from the theoretical indicated work of the expansion cycle W_{ie} (Eq. (14)), is lower than in the case of a Joule cycle.

The thermodynamic efficiency (Eq. (21)) is higher with a Joule cycle, although the theoretical specific indicated work is lower, because of the higher quantity of effective heat required for air transformations with the Ericsson cycle. Indeed, for an Ericsson compression cycle, the air entering the heat exchanger is at lower temperature than for a Joule compression cycle (isothermal versus isentropic transformations), inducing a higher heat transfer at this step, as expressed in Eq. (13) with the term $Q_{eff,H}$. An other heat transfer occurs also, when compared to the Joule cycle, during the expansion of the residual mass in the compression cylinder: it corresponds to the term $Q_{eff,C}$ in Eq. (13).

A compression cylinder working with a Joule cycle is more favourable than an Ericsson one regarding the investment costs (compression cylinder volume smaller) and the engine thermodynamic efficiency, despite the higher theoretical indicated work of the compression cylinder W_{ic} . So the compression cylinder should be insulated to avoid heat transfers in the cylinder wall. The Joule cycle is close to the real compression cycle because the rapid air mass flow rate prevents the heat transfers through the cylinder wall, which facilitates its implementation in the compression cylinder.

4.3. Influence of the expansion cycle shape

The influence of the expansion cycle shape is studied by considering a Joule cycle in the compression cylinder. The

expansion cycle shape can include an incomplete expansion (step (2_e–3_e–3_e' in Fig. 3) or a residual mass compression (step (0_e–0_e' in Fig. 3). Compression and expansion transformations in the expansion cylinder can be isentropic (Joule cycle) or isothermal (Ericsson cycle).

4.3.1. Influence of the incomplete expansion

The incomplete expansion transformation of the expansion cycle (cycle (0_e–1_e–2_e'–3_e'–3_e'') in Fig. 3, considered without residual mass compression), induced by a late intake valve closing, can be adjusted in the model via a coefficient called α_{in} . It is defined as the proportion between the volume swept during the air intake (step (1_e–2_e) in Fig. 3) and the expansion cylinder swept volume, so that its value is included between the value for a complete expansion (step (2_e–3_e) in Fig. 3) and 1.

$$\alpha_{in} = \frac{V_{2'_e} - V_{me}}{V_{me} - V_{me}} \quad (25)$$

Sometimes, the maximum value of α_{in} is lower than 1 to dismiss the non-functioning points, for example when the theoretical indicated work of the compression cylinder W_{ic} becomes higher than the absolute value of the theoretical indicated work of the expansion cylinder $|W_{ie}|$ (Eq. (14)).

Several tests are performed to determine the influence of an expansion cycle with incomplete expansion on the engine performances, for different maximum pressures (cf. Figs. 4 and 5). A Joule compression cycle is considered in the results presented. Fig. 4 plots the indicated mean pressure IMP, defined in Eq. (16), and the thermodynamic efficiency η_{th} , defined in Eq. (21). Fig. 5 plots the theoretical specific indicated work w_i , described in Eq. (15) and the adimensional mass flow rate q_m^* as a function of the coefficient α_{in} , for both Joule (J) and Ericsson (E) expansion cycles. The adimensional mass flow rate is defined as:

$$q_m^* = q_m / q_{m,ref} \quad (26)$$

The reference value of air mass flow rate $q_{m,ref}$ is evaluated from Eq. (18) with m_{cycle} chosen as the equivalent air mass present in the expansion cylinder swept volume at atmospheric pressure and temperature.

The pressure at the heat exchanger outlet p_h influences the minimum value of α_{in} (value for a complete expansion (2_e–3_e) in Fig. 3): the higher the pressure is, the lower is the minimum value

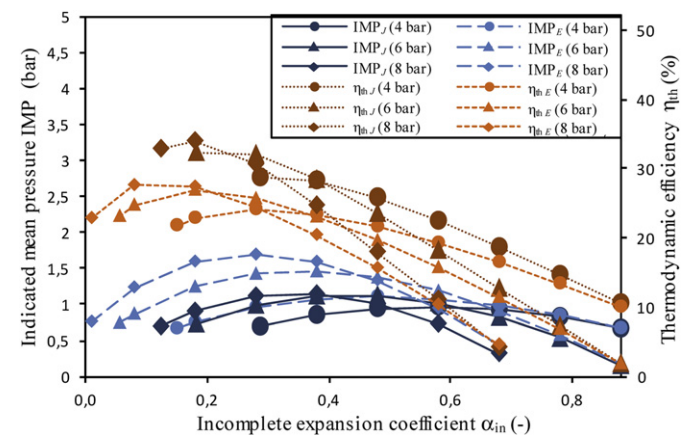


Fig. 4. Evolution of the indicated mean pressure IMP and the thermodynamic efficiency η_{th} versus the α_{in} coefficient of an Ericsson engine working with an Ericsson (E) or Joule (J) expansion cycle under several pressures p_h (4, 6 or 8 bar).

Table 3

Engine global performances obtained with Joule and Ericsson compression cycles.

Joule	Ericsson
$\tau_c = 11.68$	$\tau_c = 14.06$
$T_{0c} = 216^\circ\text{C}$	$T_{0c} = 20^\circ\text{C}$
$w_i = 107.5 \text{ kJ/kg}$	$w_i = 153.4 \text{ kJ/kg}$
$Q_{eff} = 332.2 \text{ kJ/kg}$	$Q_{eff} = 527.7 \text{ kJ/kg}$
$\eta_{th} = 32.36\%$	$\eta_{th} = 29.06\%$

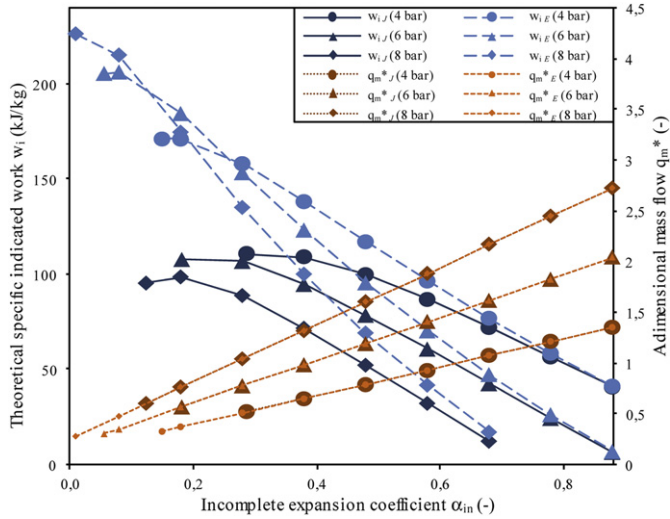


Fig. 5. Evolution of the theoretical specific indicated work w_i and the adimensional mass flow rate q_m^* versus the α_{in} coefficient of an Ericsson engine working with an Ericsson (E) or Joule (J) expansion cycle under several pressures p_h (4, 6 or 8 bar).

of α_{in} (cf. Figs. 4 and 5) i.e. the lower is the volume at the point 2_e . This is caused by the decreasing curves of isentropic and isothermal expansion in the P–V diagram: for a fixed isentropic curve (determined by the expansion cylinder maximum volume V_{Me}), a higher pressure is linked to a lower volume.

At a fixed pressure p_h , the increase of α_{in} induces the increase of the air mass flow rate (cf. Fig. 5). This is explained by an induced intake air volume ($V_{2_e} - V_{1_e}$) higher and so a higher air intake mass per cycle. Since the engine rotational speed is supposed constant in the model, this leads to an increase of the air mass flow rate.

The indicated mean pressure reaches a maximum value for both Ericsson and Joule expansion cycles when α_{in} increases (cf. Fig. 4). This is linked to a maximal value of the theoretical indicated work W_i defined in Eq. (14), which is due to the simultaneous increase of the absolute value of the expansion cycle indicated work $|W_{ie}|$ and of the compression cycle indicated work W_{ic} when the air mass flow rate increases. The first one increases more than the second one for low values of α_{in} and increases more slowly for high values of α_{in} , inducing an optimum value of α_{in} regarding the indicated mean pressure.

The theoretical specific indicated work w_i reaches a maximum value for the same reasons as the indicated mean pressure, but the maximum value is reached for a lower value of α_{in} because of the increasing of the intake air mass which influences the theoretical specific indicated work (cf. Eq. (15)). In the same way, the thermodynamic efficiency reaches a maximum value for a lower α_{in} value than in the case of the indicated mean pressure. This is due to the effect of the heat transfer increasing with the air mass flow rate passing through the engine, which is a consequence of the α_{in} coefficient increase.

The curves of indicated mean pressure, theoretical specific indicated work and thermodynamic efficiency are sometimes interrupted at high α_{in} values, when the compression cycle indicated work W_{ic} becomes higher than the expansion cycle indicated work W_{ie} .

The engine performances evolutions versus the α_{in} coefficient are similar for a Joule or an Ericsson expansion cycle. For a fixed pressure, the air mass flow rates are similar, the maximum indicated mean pressure and the theoretical specific indicated work are higher, and the maximum thermodynamic efficiency is lower for an Ericsson expansion cycle. According to these results, the heat

transfers in the expansion cylinder wall should be enhanced for an optimized indicated mean pressure and an optimized specific indicated work, or should be avoided to improve the thermodynamic efficiency.

To optimize the performances regarding the expansion cycle shape with an incomplete expansion, a choice on the mean variable (indicated mean pressure, theoretical specific indicated work or efficiency) or a compromise between the optimal α_{in} values have to be made. The values of α_{in} used in this study are the values in the case of a complete expansion (2_e-3_e) (cf. Fig. 3) or the values maximizing the thermodynamic efficiency.

4.3.2. Influence of the residual mass compression

With a similar reasoning than previously during the air intake, we can define a coefficient α_{EEVC} during the residual mass compression of the expansion cycle (step $0'_e-0''_e$ in Fig. 3) induced by an early exhaust valve closing (EEVC):

$$\alpha_{EEVC} = (V_{0'_e} - V_{me}) / (V_{Me} - V_{me}) \quad (27)$$

The values of the coefficient α_{EEVC} are comprised between 0 and the value for a residual mass compression reaching the maximum pressure p_h .

Figs. 6 and 7 show the engine performances depending on the α_{EEVC} coefficient under several maximum pressures p_h : the indicated mean pressure IMP, the thermodynamic efficiency η_{th} (cf. Fig. 6), and the theoretical specific indicated work w_i , the adimensional mass flow rate q_m^* (cf. Fig. 7), for a cycle with complete expansion (cycle $0'_e-0''_e-1_e-2_e-3_e$ in Fig. 3). The temperature differences ($T_{ie}-T_h$), with i matching the points (1_e) or (2_e) (cf. Fig. 3) depending on the α_{EEVC} coefficient for several pressures p_h are presented in Fig. 8. The results are obtained with a compression cycle considered as a Joule cycle for both Joule and Ericsson expansion cycles.

The maximum value of α_{EEVC} is higher when the pressure p_h increases (cf. Figs. 6 and 7) because of the shift in isentropic or isothermal transformations during the residual mass compression in the P–V diagram when the pressure changes.

At a fixed pressure p_h , the air mass flow rate decreases with α_{EEVC} for both Joule and Ericsson expansion cycles (Fig. 7) because the residual mass compression from a higher volume (or an earlier exhaust valve closing) induces the presence of a higher residual mass in the cylinder, which reduces the intake air mass. Indeed, when α_{EEVC} increases, the step ($0'_e-1_e$) in Fig. 3 is reduced (lower

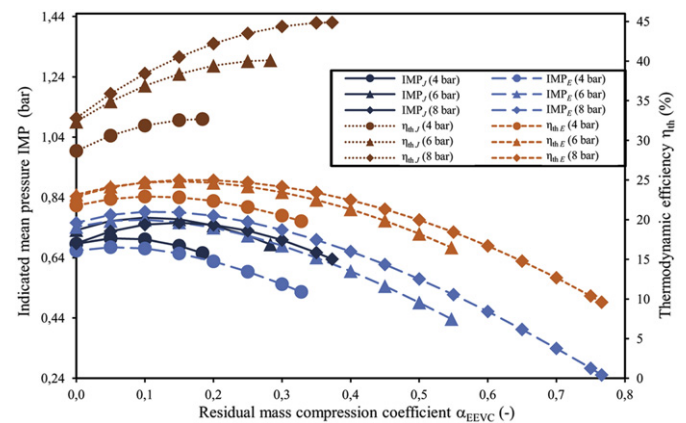


Fig. 6. Evolution of the indicated mean pressure IMP and the thermodynamic efficiency η_{th} versus the coefficient α_{EEVC} of an Ericsson engine working with an Ericsson (E) or Joule (J) expansion cycle under several pressures p_h (4, 6 or 8 bar).

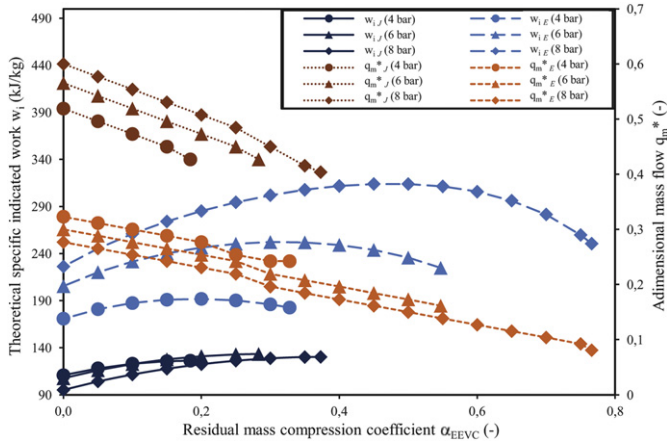


Fig. 7. Evolution of the theoretical specific indicated work w_i and the adimensional mass flow rate q_m^* versus the coefficient α_{EEVC} of an Ericsson engine working with an Ericsson (E) or Joule (J) expansion cycle under several pressures p_h (4, 6 or 8 bar).

pressure difference), so the intake air mass entering in the expansion cylinder during the pressure balance decreases, which reduces the air mass flow rate.

The pressure balance ($0'_e-1_e$) (cf. Fig. 3) induces a temperature T_{1e} higher than the intake air temperature of the expansion cylinder T_h for both Joule and Ericsson expansion cycles (cf. Fig. 8) because of the successive adiabatic air compressions at constant volume in the cylinder during the pressure balance ($0'_e-1_e$) in Fig. 3 (cf. section 3.2.3). The higher the pressure difference ($p_{1e}-p_{0e}$) is (which corresponds to a lower α_{EEVC}), the higher is the temperature T_{1e} . The increase of the temperature T_{1e} induces an increase of the temperature T_{2e} because of the mixing between the intake air and the air present in the cylinder at the point 1_e during the air intake phase (1_e-2_e) (cf. Fig. 3).

A maximum value of the indicated mean pressure depending on α_{EEVC} is observed for Joule and Ericsson cycles in Fig. 6. This is the result of the simultaneous variations of the compression and expansion cycles indicated works which decrease differently when α_{EEVC} increases.

The thermodynamic efficiency η_{th} (cf. Fig. 6) and the theoretical specific indicated work w_i (cf. Fig. 7) depending on α_{EEVC} reach maximum values in the domain of functioning of the engine for both Joule and Ericsson expansion cycles. These maximum values

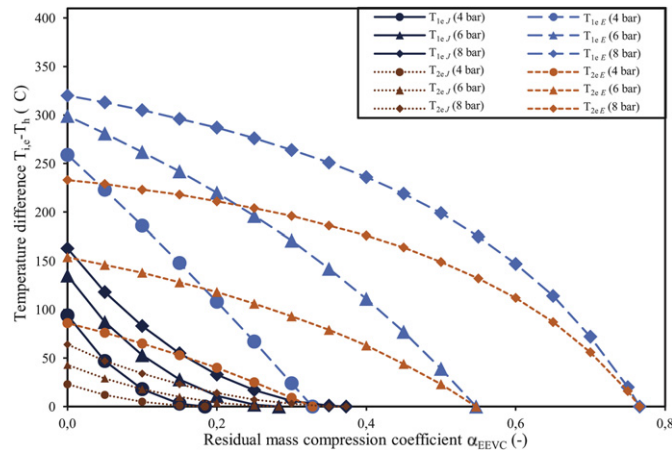


Fig. 8. Evolution of the temperature difference ($T_{1e}-T_h$), with $T_{1e} = T_{1e}$ or $T_{1e} = T_{2e}$, versus the coefficient α_{EEVC} of an Ericsson engine working with an Ericsson (E) or Joule (J) expansion cycle under several pressures p_h (4, 6 or 8 bar).

are reached for higher α_{EEVC} values than in the case of the indicated mean pressure because of the influence of the air mass flow rate on the thermodynamic efficiency and on the theoretical specific indicated work. Indeed, when α_{EEVC} increases, the intake air mass per cycle decreases, as well as the air mass flow rate, so that the heat transferred decreases. These last variables have a strong influence on the thermodynamic efficiency (Eq. (21)) and on the theoretical specific indicated work (Eq. (15)).

The engine performances with Joule and Ericsson expansion cycle present similar behaviours in the domain of functioning of the engine. The thermodynamic efficiency and the adimensional mass flow rate are higher, and the theoretical specific indicated work is lower for the Joule expansion cycle, while the indicated mean pressures are at similar levels. For the proposed application and in the range of air pressures and temperatures at the heat exchanger outlet, the Joule cycle is more adapted than the Ericsson cycle to enhance the engine performances, in particular to improve the thermodynamic efficiency.

The optimum α_{EEVC} coefficient can be chosen to enhance either the thermodynamic efficiency, or the theoretical specific indicated work, or the indicated mean pressure. It can also be a compromise between the three optimum values of this coefficient. The α_{EEVC} coefficient values chosen in the following paragraphs maximize the thermodynamic efficiency.

4.4. Influence of the pressure p_h and the temperature T_h

The influence of the pressure p_h and the temperature T_h on the engine performances is studied for the expansion cycle ($0'_e-0''_e-1_e-2'_e-3'_e-3''_e$) (cf. Fig. 3), adjusted with the optimal values of α_{in} (cf. Eq. (25)) and α_{EEVC} (cf. Eq. (27)) enhancing the thermodynamic efficiency. The values chosen for α_{in} and α_{EEVC} can also be focused on the improvement of the indicated mean pressure, leading to an expansion cycle shape different of the one maximizing the thermodynamic efficiency taken in this section. The compression cycle considered here is a Joule cycle for both Joule and Ericsson expansion cycles. Fig. 9 shows the indicated mean pressure and the thermodynamic efficiency depending on the pressure p_h , under several values of the temperature T_h . Fig. 10 shows the theoretical specific indicated work and the adimensional mass flow rate depending on the pressure p_h , under several values of the temperature T_h .

An increase of the temperature T_h leads to an increase of the indicated mean pressure, the thermodynamic efficiency (cf. Fig. 9) and the theoretical specific indicated work (cf. Fig. 10). Indeed, for a fixed indicated work of the compression cycle, the indicated work of the expansion cycle increases, which enhances the engine performances.

The thermodynamic efficiency is improved for high values of the pressure p_h because the absolute value of the expansion cycle indicated work $|W_{ie}|$ increases faster with the pressure than the compression cycle indicated work W_{ic} (cf. Eq. (14)). However, when the thermodynamic efficiency is high, the air mass flow rate and the indicated mean pressure are low, so that the global engine performances are not optimal.

In the case of a Joule expansion cycle with a complete expansion and a maximum residual mass compression, when the points $0''_e$ and 1_e in Fig. 3 are the same, the thermodynamic efficiency is independent of the temperature T_h :

$$\eta_{th} = 1 - \left(\frac{p_{atm}}{p_h} \right)^{\frac{\gamma-1}{\gamma}} \quad (28)$$

The indicated mean pressure and the adimensional mass flow rate depending on the pressure p_h reach an optimum value which is

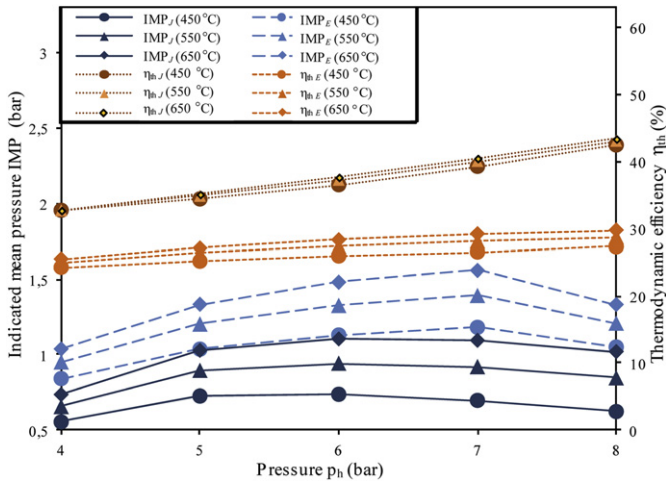


Fig. 9. Evolution of the indicated mean pressure IMP and the thermodynamic efficiency η_{th} versus the pressure p_h of an Ericsson engine working with an Ericsson (E) or Joule (J) expansion cycle under several temperatures T_h (450 °C, 550 °C and 650 °C).

at the same pressure when the temperature T_h is changed. Indeed, in the model, this temperature is not linked with the pressure p_h , or the indicated works of the compression and expansion cycles W_{ic} and W_{ie} .

For a Joule expansion cycle, the specific indicated work remains stable with the pressure p_h variations. It rises with the pressure p_h for an Ericsson expansion cycle. These results are obtained with the case of an expansion cycle shape optimizing the thermodynamic efficiency (specific values of α_{in} and α_{EEVC} chosen).

5. Discussion

Table 4 shows the working conditions and performances of an Ericsson engine allowing a satisfactory compromise between high values of indicated mean pressure (over 1 bar to overtake the maximum brake effective pressure) and thermodynamic efficiency. The compression and expansion cycles chosen are Joule cycles (insulated cylinder walls), which enhance the thermodynamic

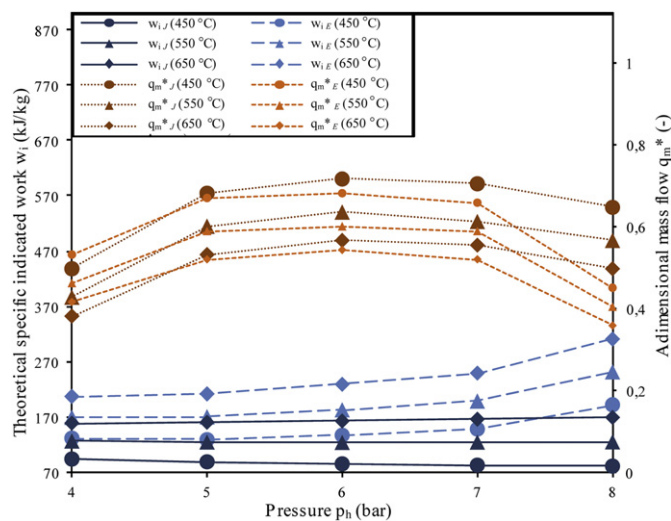


Fig. 10. Evolution of the theoretical specific indicated work w_i and the adimensional mass flow rate q_m^* versus the pressure p_h of an Ericsson engine working with an Ericsson (E) or Joule (J) expansion cycle under several temperatures T_h (450 °C, 550 °C and 650 °C).

efficiency. The pressure p_h is fixed at 6 bar to maximize the indicated mean pressure and the temperature T_h equals 650 °C (the highest value reached with the existing heat exchangers) to improve simultaneously the indicated mean pressure, the thermodynamic efficiency and the theoretical specific indicated work. The expansion cycle shape is adjusted by means of the coefficients α_{in} and α_{EEVC} to maximize the thermodynamic efficiency. This leads to performances which are a compromise between the highest values of thermodynamic efficiency, indicated mean pressure and theoretical specific indicated work. The exergetic efficiency η_{ex} presented in Table 4 and the comparison of the exergy flow rate supplied by the heat exchanger $\dot{Ex}_H$ with the exergy flow rate rejected in the atmosphere $\dot{Ex}_{out}$ show the suitability of the Joule cycle Ericsson engine for the energy conversion, in accordance with the results obtained by Bonnet et al. [8].

This study of the influent parameters on the global performances of the Ericsson engine (compression and expansion cylinders coupled) leads to an optimization of the working conditions. This approach allows the definition of a range of optimized working parameter values, which will simplify the next step of the engine optimization with a dynamic model.

Some dynamic effects could be evaluated in our model, like the pressure drops across the valves and the pressure drop in the heat exchanger, or the frictional effects.

Although the model presented here does not include the frictional effects in the cylinders, this is a very important aspect. Thus, the indicated mean pressure must be specially considered: its value has to be higher than the friction mean pressure (maximum value of 1 bar) to ensure the functioning of the engine. For this purpose, if the value of the indicated mean pressure is too low, an optimization (similar to the one presented here which is focused on thermodynamic efficiency improvement) can be established to enhance preferentially the indicated mean pressure.

An extension of this study could take into account a regenerator. This system reduces the required heat transferred to the working air in the heating phase, enhancing the theoretical global mechanical efficiency of the engine, which does not change the results of this study (the optimum working conditions are the same). In the same way, the efficiency of the heat exchanger could be added in the model to evaluate the real heat quantity required by the facility and to deduce the cogeneration efficiency of the engine.

In order to compare the interest of using an Ericsson engine for the biomass energy upgrading with the Stirling engine (already used for this application), the working conditions of the Stirling engine have to be similar to the Ericsson engine (pressure below 10 bar and temperature below 800 °C). Several studies found in the literature allow this comparison: Cinar et al. investigated a β -type [18] and a γ -type [19] Stirling engine working at atmospheric

Table 4

Performances and working conditions of an Ericsson engine optimized (Joule compression and expansion cycles).

Working conditions	Performances
$\alpha_{in} = 0.3$	IMP = 1.104 bar
$\alpha_{EEVC} = 0.283$	$T = 12.6$ Nm
$p_h = 6$ bar	$q_m = 4.9$ g/s
$T_h = 650$ °C	$P_{th} = 2111.8$ W
$\tau_e = 10$	$P_i = 795$ W
$\tau_c = 11.7$	$\eta_{th} = 37.6\%$
$N = 600$ rpm	$m_{cycle} = 0.489$ g/cycle
	$Q_{eff} = 211$ J/cycle
	$W_i = 79$ J/cycle
	$T_{out} = 290$ °C
	$\dot{Ex}_H = 1119$ W
	$\dot{Ex}_{out} = 342$ W
	$\eta_{ex} = 71.0\%$

pressure, and Kongtragool et al. [20] described a Stirling engine with working conditions adapted to the Ericsson engine.

The parametric optimization presented allows to precise the functioning domain of the Ericsson engine (adjustment of the working conditions), with a focus on thermodynamic aspects. Further investigations have to be performed to include the other aspects of the Ericsson engine design: mechanical effects, dynamic behaviour.

6. Conclusion

The functioning of the Ericsson engine with Joule or Ericsson compression and expansion cycles has been investigated in a global approach to optimize the engine performances. The basic engine configuration, with a compression cylinder, an expansion cylinder and a heat exchanger, is used to separate the performances of the engine from the performances of a cogeneration system based on an Ericsson engine where the heat at the outlet is reused, for example in a regenerator or a pre-heater (some cases are already described in the literature [11–13]).

The results based on a thermodynamic model in a steady state are presented. This allows the analysis of the influencing parameters on the engine performances: heat transfers through the compression and expansion cylinder walls, expansion cycle shape, pressure and temperature at the outlet of the heat exchanger.

A parametric optimization of the engine performances (thermodynamic efficiency, theoretical specific indicated work and indicated mean pressure) is set up. The optimum range of values of the influent parameters are determined thanks to a compromise ensuring the highest possible thermodynamic efficiency of the engine with a satisfactory indicated mean pressure (over 1 bar to overtake the maximum possible friction mean pressure). The optimized values of the theoretical specific indicated work are a consequence of the optimal values obtained for the indicated mean pressure. The optimization of the engine functioning can be improved: in a next step, a dynamic model of the engine will be set up, including the frictional effects, the pressure drops in the heat exchanger and through the valves, and a detailed exergetic analysis of the engine. The performances are optimum for an Ericsson engine working with Joule compression and expansion cycles (no heat transfers in the cylinder walls), under a pressure between 5 and 8 bar, and at the highest possible temperature that the heat exchanger can reach. The expansion cycle shape has to be adjusted with a residual mass compression (early exhaust valve closing) and an incomplete expansion (late intake valve closing).

Acknowledgement

Ms. Creyxx was supported by a joint PhD grant from the company Enerbiom and the ANRT (National Association of Research and Technology).

This work was carried out within the framework of the regional project SYLWATT, funded by the Région Nord-Pas-de-Calais, in partnership with TEMPO Valenciennes, PC2A Lille, CCM Dunkerque and Enerbiom.

References

- [1] Danon G, Furtula M, Mandić M. Possibilities of implementation of CHP (combined heat and power) in the wood industry in Serbia. *Energy* 2012;1–8.
- [2] Houwing M, Ajah AN, Heijnen PW, Bouwmans I, Herder PM. Uncertainties in the design and operation of distributed energy resources: the case of micro-CHP systems. *Energy* 2008;33:1518–36.
- [3] Casisi M, Pinamonti P, Reini M. Optimal lay-out and operation of combined heat & power (CHP) distributed generation systems. *Energy* 2009;34:2175–83.
- [4] Roy JP, Mishra MK, Misra A. Performance analysis of an Organic Rankine Cycle with superheating under different heat source temperature conditions. *Energy* 2011;88:2995–3004.
- [5] Biedermann F, Carlsen H, Schöch M, Obernberger I. Operating experiences with a small-scale CHP pilot plant based on a 35 kW_{el} hermetic four cylinder stirling engine for biomass fuels. In: Proc. of the 2nd int. conf. on structural and construction engineering, Rome, Italy; September 2003. p. 23–6.
- [6] Thiers S, Aoun B, Peuportier B. Experimental characterization, modeling and simulation of a wood pellet micro-combined heat and power unit used as a heat source for a residential building. *Energy and Building* 2010;42: 896–903.
- [7] Podesser E. Electricity production in rural villages with a biomass stirling engine. *Renewable Energy* 1999;16:1042–52.
- [8] Bonnet S, Alaphilippe M, Stouffs P. Energy, exergy and cost analysis of a micro-cogeneration system based on an Ericsson engine. *International Journal of Thermal Sciences* 2005;44:1161–8.
- [9] Costea M, Feidt M. The effect of the overall heat transfer coefficient variation on the optimal distribution of the heat transfer surface conductance or area in a Stirling engine. *Energy Conversion and Management* 1998;39:1753–61.
- [10] Kaushik SC, Kumar S. Finite time thermodynamic evaluation of irreversible Ericsson and Stirling heat engines. *Energy Conversion and Management* 2001; 42:295–312.
- [11] Wojewoda J, Kazimierski Z. Numerical model and investigations of the externally heated valve Joule engine. *Energy* 2010;35:2099–108.
- [12] Moss RW, Roskilly AP, Nanda SK. Reciprocating Joule-cycle engine for domestic CHP systems. *Applied Energy* 2005;80:169–85.
- [13] Bell MA, Partridge T. Thermodynamic design of a reciprocating Joule cycle engine. Proceedings of the Institution of Mechanical Engineers, Part H, Journal of Power and Energy 2003;217:239–46.
- [14] Lontsi F, Castaing-Lavignottes J, Stouffs P. Dynamic simulation of a small Joule cycle Ericsson engine: first results. In: Proc. of the int. stirling forum 2008. Osnabrück, Germany: ECOS GmbH; 2008.
- [15] Touré A. Etude théorique et expérimentale d'un moteur Ericsson à cycle de Joule pour conversion thermodynamique d'énergie solaire ou pour micro-cogénération. PhD thesis, Université de Pau et des Pays de l'Adour, France; 2010.
- [16] Chen J, Schouten JA. The comprehensive influence of several major irreversibilities on the performance of an Ericsson heat engine. *Applied Thermal Engineering* 1999;19:555–64.
- [17] Benelmir R, Lallemand A, Feidt M. Analyse exergetique, Techniques de l'Ingenieur, BE8015; 2002. p. 1–15.
- [18] Cinar C, Yucsu S, Topgul T, Okur M. Beta-type Stirling engine operating at atmospheric pressure. *Applied Energy* 2005;81:351–7.
- [19] Cinar C, Karabulut H. Manufacturing and testing of a gamma type stirling engine. *Renewable Energy* 2005;30:57–66.
- [20] Kongtragool B, Wongwises S. A review of solar-powered Stirling engines and low temperature differential Stirling engines. *Renewable and Sustainable Energy Review* 2003;7:131–54.